3. NOISE EMISSION AND CONTROL

The key to reducing noise of gas flows is to understand the mechanisms by which noise is produced and transmitted to the environment, and to design for the opposite result to whatever extent possible.

3.1. General Discussion

Noise is a waste byproduct of mechanical processes. A very small fraction of the mechanical energy in a given process reaches our ears as sound. The fraction is small primarily because of various inefficiencies in converting mechanical energy into acoustic energy.

For most mechanical equipment, casing vibration creates waves radiating into the atmosphere with an efficiency ranging from 10^{-5} to 10^{-7}. In other words, it may take as much as one megawatt of mechanical power to produce one acoustic watt. While that may at first seem encouraging, one acoustic watt is a rather large quantity that is capable of causing hearing damage to personnel nearby.

Gas flow systems are potentially more noisy than mechanical equipment, however, because the mechanical power is already part of the gas flow: there is considerably less mechanical/acoustical conversion inefficiency to overcome. Gas flow systems convert their mechanical power to acoustical power at efficiency rates ranging from 10^{-3} to 10^{-5}. This is especially problematic when the gas flow is not contained within piping but comes into direct contact with the atmosphere. High velocity gas discharge vents and aircraft engine exhausts are cases in point.

For perspective, it is worth noting that loudspeakers and other similar devices specifically designed to radiate sound do so with efficiency of approximately 10^{-2}, or 1%.

Let us summarize the above using W_M for stream mechanical power at the point of noise generation, W_A for acoustic power and η for efficiency, and substitute for W_M:

$$W_A = \eta W_M$$
$$W_A = \eta FV$$

Assuming that the force acts over the same area the flow passes through, this can be simplified further to

$$W_A = \eta \Delta P Q$$

It should be clear from this simplified approach that in order to reduce noise output, three primary options are available:

- Reduce efficiency of conversion to acoustic power,
- Reduce force exerted on the gas by reducing either the pressure differential or the area over which it acts,

- Reduce the velocity of the gas by reducing the volume flow or increasing the flow area.

A fourth important option is not obvious from the above list:

- Modify the design to cause energy to be expressed frequency bands less likely to cause hearing damage.

Judicious application of these four approaches is the key to successful noise reduction in gas flows.

3.2. Sound Power Level and Sound Pressure Level

It is important to properly understand the distinction between sound power and sound pressure. The acoustic power of a source in watts is called the *sound power*. This quantity represents the energy output of the source per unit time into its environment. *Sound power level* is a decibel expression of the sound power referenced to 10^{-12} watts:

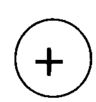
$$L_W = 10\log_{10}\frac{W_A}{10^{-12}\,watt}$$

Sound pressure is the expression of that energy filling the environment, just as temperature is the expression of thermal energy filling the environment. In the case of heat, it is clear that the temperature in a heated space is a function not only of the power of the heater, but also on the proximity of the observer, the ability of the environment to contain heat, and the ambient temperature that would prevail independent of the heater.

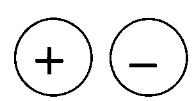
3.3. Noise Generation in Gas Flows

Three types of acoustic sources are responsible for most of the noise in gas flows: monopoles, dipoles and quadrupoles.

A *Monopole* is the simplest type of source, corresponding to a pulsation of gas pressure or velocity. A monopole source is like a pulsating sphere. Pressure or velocity pulsations are in phase at all points on the source. A vibrating duct wall or open end of a pipe might serve as a monopole under certain conditions.

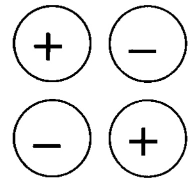


Dipoles are the consequence of oscillatory forces in the flow arising chiefly from interactions between the gas and structures. A dipole is analogous to two monopole



sources oscillating out of phase and separated by a small distance (compared to a wavelength). Compressor blades give rise to dipoles because they have high pressure on one side and low pressure on the other. Dipoles also are found in the periodically alternating vortices shed from flow obstructions such as struts. In many cases, dipoles lead to tonal (pitched) components in the noise.

Quadrupoles result from viscous stresses in turbulent flow in the absence of obstacles. A quadrupole is analogous to two dipoles that oscillate out of phase and separated by a small distance (compared to a wavelength). Wherever turbulence and/or mean velocity gradient are high, quadrupole source strength may be significant.



For a given mechanical power and size compared to an acoustic wavelength, a monopole is the most efficient radiator of sound, followed by dipoles and quadrupoles. The amount of sound energy radiated, however, is proportional to u^4, u^6 and u^8, respectively, where u is a local flow perturbation (acoustic) velocity. Thus at high velocities (on the order of Mach 1) the quadrupole source strength can predominate over dipoles and monopoles.

In general, noise control is most effective when all three types of acoustic sources are minimized. However, where one type is predominant, minimizing the conditions that give rise to that particular type of acoustic source is usually the most successful approach.

3.4. Noise Emission

Noise is emitted from the gas flow system in one of three ways:

- by radiation from a gas flow boundary where the noise is produced, such as for a high velocity unconstrained gas jet.
- by radiation from pipe and duct walls, which vibrate in response to fluctuating pressures due to turbulent processes or acoustic excitation within,
- by radiation of noise within the piping system from an intake or discharge opening in the system.

3.4.1. Noise Emission from a Flow Boundary

The magnitude of noise emitted at a gas flow shear boundary depends chiefly on the velocity of the gas jet relative to the ambient atmosphere, but also on the nozzle area and on the density and temperature of the jet relative to the atmosphere. Sound generated within the gas jet core is refracted on passing through the shear layer. More detail is given in Chapter 1 and in the references given.

3.4.2. Noise Emission from Piping

Noise generated within a piping system (e.g., by a compressor) propagates in both the upstream and downstream directions if the flow is subsonic. As the flow nears sonic velocity, most of the energy travels in the downstream direction. Above sonic velocity, all of the sound energy is convected downstream with the flow.

As sound energy travels through the piping, a small fraction is expended in vibrating the pipe or duct walls, which in turn re-radiate the sound to the environment. The remainder of the sound energy usually remains within the gas. This is beneficial on one hand because sound levels outside the piping are reduced. On the other hand, the sound energy trapped within the pipe travels great distances, often without significant attenuation. When the sound energy emerges at remote locations, unintended noise emission problems can arise.

No significant loss of acoustic energy should be expected along the first several hundred diameters of round piping length. The System Analysis model in the Workbook assumes no acoustic loss other than sound transmission along the length of any pipe.

Higher values of pipe- or duct wall *transmission loss* indicate lesser fractions of sound transmitted. It turns out that circular pipe has high transmission loss in all but a small band of frequencies, and sound levels decay only very slowly with distance (a fraction of a dB per 100 diameters). By contrast, rectangular duct profiles have significantly lower transmission loss at low frequencies, and release a greater proportion of sound to the environment. However, in that case the sound levels decay more rapidly with distance.

Methods of reducing noise emission from piping are discussed in Chapter 8.

3.4.3. Transmission from Open Duct End

Sound propagating within a piping system may eventually reach an intake or discharge opening. An abrupt acoustic impedance change at the open end causes some waves (particularly at low frequency) to be unable to exit the opening. This effect is increased by significant inflow and decreased by significant outflow.

Horn-like structures at the end of the duct may actually increase sound radiation by diminishing the impedance change.

3.4.4. Radiation to Environment

Within the *near field* of a source of sound (within approximately one source dimension) the sound pressure level fluctuates considerably but on the whole does not decay with distance. In the far field (several source dimensions distant), the sound pressure level decreases approximately 6 dB per doubling of distance as long as there are no reflecting surfaces (other than the ground) present. At greater distances outdoors, levels may decrease more rapidly because of atmospheric and ground effects. Practical control of the received level outdoors can only be achieved by reducing the level of the source, or by erecting a barrier or enclosure close to either the source or receiving point.

When multiple sources of noise are present, the sound energies produced are additive. In such a case sound pressure levels will generally be higher (by as much as 5 dB(A)) than the highest sound pressure level produced by any one piece of equipment. For this reason, noise control efforts must begin with the equipment producing the highest sound pressure level and can only be expected to reduce levels to those produced by the equipment not treated. For example, suppose two machines each produce 85 dB(A) at a given location. The combined level would be 88 dB(A). If noise control were applied to reduce one source from 85 dB(A) to 65 dB(A), the combined level would be 85 dB(A). Thus, a 20 dB(A) noise control treatment yielded in this case a net benefit of 3 dB(A). (See Appendix C for details on decibel mathematics.)

In an indoor environment, reflected sound tends to build up so that sound levels decay less rapidly with distance, reaching an approximately constant level. Increasing the surface area covered with sound absorbing material can reduce the reverberant level. This is especially important when multiple equipment items are present: the reverberant sound pressure levels from individual equipment items are additive. A reverberant space causes otherwise "local" noise emission challenges to become "global" ones that may effect may locations and employees with a building.

4. REDUCED-NOISE DESIGN FOR FREE JETS

A free jet is defined for the purposes of the *Design Guide* as an unimpeded discharge of high velocity gas into the atmosphere. Free jets include gas and steam discharge to atmosphere. The "jet" in question is of an industrial character, wasting its thrust as it escapes (in most cases) from the open end of a pipe. Where a more formal nozzle is used that is intended to maximize thrust, the discussion of aircraft jet engine mixing and shock-associated noise in Section 7.4, (page 7-3) may be more relevant.

Note that the jet formed by an intake (vacuum) vent is not free but constrained within downstream piping or a vessel. Intake vents are discussed in Section 5.2.3, page 5-5.

4.1. Mechanism of Noise Production for Free Jets

High velocity gas interacts with the surrounding atmosphere at rest to produce significant shear stresses and turbulent mixing. This mixing produces sound. The overall sound power output W_A of the jet is taken to be dependent on the eighth power of exit velocity U_j after Lighthill[9]:

$$W_A \propto \frac{\rho_j S_j U_j^8}{c^5}$$

where

ρ_j is the jet density,
S_j is the fully expanded jet area, and
c is the sonic velocity in ambient air.

Small-scale vortices give rise to high frequency quadrupole sound sources. Larger scale vortices within the jet produce low frequency quadrupole sound sources. The frequency at which peak sound pressure occurs is approximately:

$$f_p = \frac{0.2U_j}{D_j}$$

where

f_p is the peak frequency in Hz, and
D_j is the fully expanded jet diameter.

When the ratio of upstream to ambient pressure P_1/P_A is greater than 1.5, sonic flow may exist in the vena contracta downstream of the outlet. If the ratio exceeds 1.89 (in air), the flow will definitely be sonic ($M_j > 1$). Once sonic flow is reached the flow cannot accelerate further without the help of a converging-diverging nozzle. If no C-D nozzle is present, *choked flow* is said to exist. Any further increase in flow comes about through an increase in density and entropy that resolves in shock waves in the downstream flow. Shock waves are efficient generators of noise and further increase noise emission.

If the exhaust stream is interrupted by any kind of obstacle, noise emission may be increased by as much as 10 dB(A).

Noise radiated from free jet mixing has a pronounced directionality that arises from convection of quadrupoles by the flow and by refraction at the shear boundary. Peak levels are reached 150° from the inlet axis (30° from the discharge axis). Noise emission from shocks is normally taken as omnidirectional.

An empirical model that takes into account the gross behavior of gas and steam jets based on upstream pressure and temperature and nozzle area is given below in Section 4.5 (page 4-5).

4.2. Gas and Steam Discharge

Gas and steam discharges are characterized by high pressure gas venting through a control valve, relief valve, burst disk, or similar opening to atmosphere. Continuous and intermittent vents are included in this definition, as are blowdown applications in which a stationary volume of gas is vented.

The applications here are industrial. Because the vented gas serves no further useful purpose, noise control options that reduce thrust are acceptable. For the case of aircraft engine components, jet exhaust is discussed separately in Section 7.4 (page 7-3).

4.3. Guidelines for Noise Control of Gas and Steam Discharges

Significant noise reduction is possible by use of a vent silencer in conjunction with a properly selected control valve.

- ***Employ a Vent Silencer***: A vent silencer consists of two stages; a diffuser basket and a dissipative silencer. The diffuser basket breaks the jet into a number of small jets, increasing the peak frequency and thus rendering the dissipative silencer more effective. Reductions of 10 db(A) to 50 dB(A) are achievable with various designs. Care should be taken that the self-noise of the silencer does not limit its performance.

- ***Use a Low-noise Valve:*** Noise generated at the control valve also propagates with the flow, but is not frequency-shifted by the diffuser basket. Thus the vent silencer is less effective against valve noise than against jet noise. Reductions of valve noise by vent silencers are on the order of 5 dB(A) to 35 dB(A). The downstream sound power output of the valve should be at least 15 dB(A) less than the unsilenced sound power output of the jet.

  Guidelines for reducing valve noise are given in Section 5.2.

Smaller noise reduction gains can be achieved using these methods:

- ***Reduce turbulence upstream of the exit:*** allow 6-10 pipe diameters of straight duct length before the exit or other impediment is reached. Noise emission can be increased by 5 dB(A) or more if turbulent flow reaches the exit. Note that control valves and support struts are examples of flow impediments that produce turbulence.

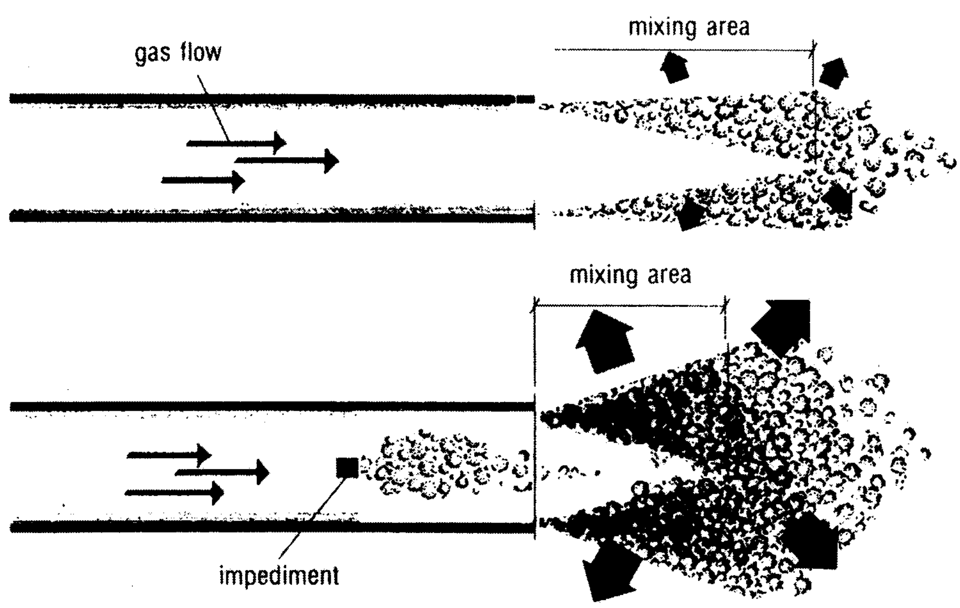


Figure 2: Effect of Turbulence Upstream of Exit

(Ingemansson and Folkesson[10])

- ***Angle of Radiation***: The axis of discharge should be oriented at least 90° away from noise sensitive areas, otherwise there is no benefit. For very large diameter outlets the benefit may be 5 dB(A) or more.

- ***Use Larger-Diameter Piping Downstream of Valve:*** The main reason for taking this action is that the dominant frequency is reduced by about 1 octave (see 4.5.2, page 4-6). The benefit of favorable directional orientation is also greater for a larger opening. In general, the net benefit is on the order of 3 dB(A) to 5 dB(A).

  A special case of this treatment is the can-type supersonic suppressor as developed by NASA GRC[11]. In this treatment, the discharge pipe is deliberately made long enough that the emerging jet boundaries strike the pipe walls. An additional 3 dB(A) to 5 dB(A) reduction may be possible.

  While abrupt area changes in flows are usually not beneficial because of increased turbulence, here the turbulence is increased so dramatically that the flow is decelerated before reaching the exit.

- ***Reduce the Pressure and Temperature of the Vented Gas,*** although this may seldom be practical. A 20% reduction in pressure yields a 1 dB(A) reduction, while a 20% reduction in gas temperature yields a 2 dB(A) noise reduction.

- ***Entrain ambient airflow*** using a co-annular eductor nozzle to reduce relative velocity in the shear layer[11]. Overall reductions of between 5 dB(A) and 10 dB(A) can be achieved using this approach. See Figure 3 below.

- ***Introduce a rotary component to the jet flow*** using radial vane structures. This works best for hot gas exhausts.[12]

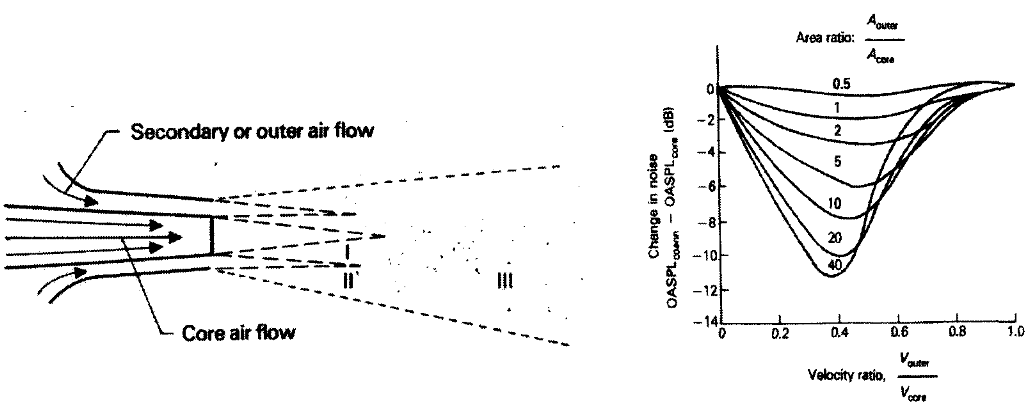


Figure 3: Effect of Entraining Airflow

(after Huff[11])

4.4. Noise Emission Estimation Using Workbook

Spreadsheets:

- Gas Vents
- Steam Vents

Required Inputs:

- Upstream conditions: pressure P_1, temperature T_1, volume V, moisture $\%m$, superheat temperature T_s
- Valve, Piping and Nozzle: valve diameter, D_v, downstream pipe diameter D_D, silencer outer diameter D_o, nozzle coefficient C_N
- Downstream conditions: pressure P_2, temperature T_2,
- Observer: distance r, angle θ

Notes:

- Reservoir volume V is an optional input
- Nozzle coefficient C_N is assumed to be 0.85 unless otherwise known.
- If no silencer is used, set silencer diameter equal to downstream pipe diameter.
- Expanded temperature and expanded density of flow must be determined from steam tables assuming that the gas has reached ambient pressure.

4.5. Predictive Equations for Discharge Vents

The Sound Pressure Level is estimated from factors for the overall sound power level, spectral shape, directivity and geometric spreading with distance.

$$SPL(f,r,\theta) = L_{W,overall} + \Delta\left(\frac{f}{f_p}\right) + D(\theta) + G(r)$$

4.5.1. Overall Sound Power Level

For air and all other gases, the emitted sound power level (dB re 1 pW) is estimated as[13]:

$$L_{W,overall} = 10\log_{10}(P_1 A_V C_N) + 20\log_{10}\left(\frac{T_1}{G}\right) + 85$$

where P_1 is pressure upstream of the control valve in psia, A_V is the valve open area in square feet, C_N is the nozzle coefficient (assumed to be 0.85 unless otherwise known), T_1 is the upstream temperature in degrees Rankine (°R) and G is the specific gravity of the gas. Downstream conditions are taken to be air at sea level, standard temperature and pressure.

For steam, the emitted sound power level (dB re 1 pW) is estimated as[13]

$$L_{W,overall} = 17\log_{10}(51.43 P_1 A_V C_N F_m F_s) + 50\log_{10} T_1 - 85$$

$$F_m = \frac{1}{1 - 0.012\,\%m}$$

$$F_s = \frac{1}{1 + 0.00065 T_s}$$

where m is the percentage moisture, and T_s is the number of degrees of superheat (°F) for superheated steam.

The peak frequency of emitted noise is

$$f_p = \frac{0.4 c_j M_j}{D_V}$$

where D_V is the valve throat diameter and c_j is the speed of sound within the gas jet at the valve exit. The speed of sound c_j can be expressed in feet per second as

$$c_j = 223\sqrt{\frac{\gamma T_j}{MW}}\ \text{ft/s}^{-1}$$

$$T_j = \frac{T}{1 + \frac{\gamma - 1}{2} M_j^{\,2}}$$

4.5.2. Spectral Shape Function Δ

The spectral shape function Δ is tabulated below as a function of the ratio of frequency to the peak frequency f/f_p. The spectral shape corrections convert the overall sound power level L_W give octave band values for two cases, here designated A and B. Case A corresponds to either no downstream piping or downstream piping the same size as the valve. Case B corresponds to downstream piping larger than the valve.

The effect of larger downstream piping after the valve is to shift the peak about one octave down in frequency. The spectral shape changes only slightly.

Table 1: Spectral Correction Factors for Gas and Steam Vents

	Frequency ratio f/f_p												
	1/28	1/64	1/32	1/16	1/8	1/4	1/2	1	2	4	8	16	32
A	-40	-36	-30	-24	-18	-12	-6	-4	-6	-12	-18	-24	-30
B	-40	-33	-22	-15	-9	-6	-5	-6	-11	-19	-29	-40	-50

The correction factors are approximated by the function:

$$\Delta|_A = -4.6 - 1.78x^2 - 3.9681\times10^{-4}x^4$$

$$\Delta|_B = -6.3 - 3.40x - 1.59x^2 + 0.1527x^3 + 0.1933x^4 + 2.7025\times10^{-4}x^5 - 1.3718\times10^{-4}x^6$$

where

$$x = \frac{\log\left(f/f_p\right)}{\log(2)}$$

4.5.3. Directivity Factor $D(\theta)$

Table 2, Table 3 and Table 4 below give directivity factors for high velocity gas or steam discharge as a function of the diameter of the outlet in inches. These values are added to the sound power level.

Table 2: Gas or Steam Discharge, 0° from Axis

	Octave Band Center Frequency [Hz]								
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
8	0	0	0	0	0	0	0	0	0
15	0	0	0	1	1	1	1	1	1
26	1	1	1	2	2	2	2	2	2
36	2	2	3	3	4	4	4	4	4
54	3	3	4	4	5	5	5	5	5
72	4	4	5	5	6	6	7	7	7

Table 3: Gas or Steam Discharge, 45° from Axis

Octave Band Center Frequency [Hz]									
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
8	0	0	0	0	0	0	0	0	0
15	0	0	0	0	0	0	0	0	0
26	0	0	0	0	1	1	1	1	1
36	0	0	1	1	2	2	2	2	2
54	1	1	2	2	3	3	3	3	3
72	2	2	3	3	4	4	5	5	5

Table 4: Gas or Steam Discharge, ≥90° from Axis

Octave Band Center Frequency [Hz]									
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	-1	-3
5	0	0	0	0	0	0	-1	-3	-6
8	0	0	0	0	0	-1	-2	-5	-11
15	0	0	0	0	0	-1	-3	-7	-13
26	0	0	0	0	-1	-3	-5	-9	-14
36	0	0	0	-1	-3	-6	-7	-11	-15
54	0	0	-1	-2	-5	-8	-10	-13	-16
72	0	-1	-2	-5	-7	-10	-12	-15	-17

4.5.4. Self Noise

A Vent Silencer (see Section 8.4, page 8-11) reduces the noise of the expanding gas flow by first converting one large jet to a large number of very small ones using a "diffuser basket". The resulting high frequency sound is then effectively absorbed as the gas flow passes between parallel baffles of sound absorbing material.

This process produces additional noise of its own called "self-noise" (Section 8.4.5, page 8-12). The self-noise sound power level is added on an energy basis to the silenced vent sound power level (in dB) to find the residual sound power level at the silencer exit.

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[9] M. J. Lighthill, On Sound Generated Aerodynamically, II., Turbulence as a Source of Sound, Proc. Roy. Soc. (London) Ser. A, vol. 222, no. 1148, Feb. 1954

[10] Stig N. P. Ingemansson, Claes Folkesson, *"Noise Control: Principles and Practice"*, this illustration from Noise News International, Vol. 3 No. 3, 1995 Sept., pp. 178-183. Published in book form by the American Society of Safety Engineers as *"Noise Control: A guide for workers and employers"*.

[11] R. H. Huff, A Simple Noise Suppressor Design for Vented High Pressure Gas, NASA Tech. Brief, summer 1979, p. 278.

[12] I. R. Schwartz, Minimization of Jet and Core Noise by Rotation of Flow, NASA Tech Brief B75-10131, 1975

[13] Bill G. Golden, Jim R. Cummins jr., "Silencer Application Handbook" , Universal Silencer, Stoughton, Wisconsin, 1993

5. CONSTRAINED JETS: CONTROL VALVES, ORIFICES, VENTURIS, VACUUM VENTS

A constrained jet is a high velocity discharge of gas into a constrained area, such as a pipe, tank or vessel. Constrained jets exist downstream of control valves, measurement orifices and venturis, and intake vents.

5.1. Mechanism of Noise Production for Constrained Jets

Constrained jets are the result of an in-line flow restriction. At the restriction the flow velocity increases and, from Bernoulli's theorem, it is known that a corresponding pressure reduction occurs. The point of maximum flow velocity and minimum static pressure is called the *vena contracta* and is located a fraction of a restricted diameter downstream.

The boundary between the fast-moving jet and slower moving gas in the pipe is the site of large shear stresses that generate small-scale vortices, with larger scale vortices created within the gas jet. The physics of the gas jet differs little from a free jet until the expanding jet contacts the walls. The difference lies in the interaction of the flow with the walls. Quadrupoles in the shear layer strike the outer wall and, along with their in-phase reflected pairs, create dipoles. The forces exerted on the pipe wall cause it to vibrate and in turn to radiate sound into the surrounding environment. Within the pipe, noise propagates through the gas and is convected with it. As sonic flow is approached, it becomes increasingly difficult for sound to travel upstream. For this reason, noise emission is often concentrated downstream of flow restrictions in control valves and on vacuum inlet vents.

No fluid is completely inviscid, so passage through the restriction incurs a pressure loss equal to $1/2K\rho U^2$ where K is a dimensionless loss factor. From Bernoulli's theorem of isentropic flows, the flow through the restriction can be shown to be:

$$Q = UA = \sqrt{P_1 - P_2} \times \sqrt{\frac{2}{\rho K}} \times A$$

where U is the mean flow velocity through the restriction or area A. The valve sizing coefficient C_V is derived from this expression as

$$C_V = \sqrt{\frac{2}{\rho_W K}} \times A$$

and assigned a numerical value for water flows expressed in gallons per minute and differential pressure in pounds per square inch, such that

$$Q = C_V \Delta P$$

Note that the value C_V/D_V^2 is a property of the valve at a particular flow condition. Actual valve sizing for real gases is more complex than can be addressed in the

Design Guide. Consult valve catalogs and sizing routines and software from control valve manufacturers.

At sonic flow speeds shock waves form and the flow is no longer isentropic. Catalog values of C_V are intended to account for all of the added complexities of the flow. Furthermore, because control valves can often be used for fluid or gas flow, catalog C_V values are often applied in gas applications. A more thorough treatment of control valve flows is contained in the literature of control valve manufacturers[14,15]

The mechanical energy in the flow is proportional to $Q\Delta P$ or $C_V\Delta P^{1.5}$. The efficiency of noise generation is proportional to the flow velocity at the point of noise generation, that is, within and just downstream of the restriction, and is noticeably increased when shocks form.

The peak flow velocity is attained in the *vena contracta*. The degree to which the *vena contracta* pressure P_0 falls below the downstream pressure P_2 is called *pressure recovery*. The *pressure recovery factor* F_L in common use for control valves is defined as

$$F_L^{\,2} = \frac{P_1 - P_2}{P_1 - P_0}$$

where P_1 is the upstream pressure. The factor F_L takes values between 0 and 1. When F_L is small, pressure recovery is complete and $P_1 - P_0 >> P_2 - P_0$. Because P_0 is less than P_2, the velocity in the *vena contracta* is higher than would be expected from the service pressure drop $P_1 - P_2$. The increased velocity corresponds to increased noise output. When $F_L \approx 1$, $P_2 = P_0$, there is no pressure recovery and the flow velocity in the *vena contracta* is essentially that in the downstream pipe. This situation usually corresponds to minimum noise output for a given pressure drop.

Confusion may result because a "high" value of F_L corresponds to low pressure recovery, and vice versa. The high value is actually preferred, because it minimizes the flow velocity in the *vena contracta* for a given pressure drop. By contrast it should be clear that the pressure drop that causes sonic flow within the restriction (and consequently high noise emission) is smaller when F_L is low than when it is high.

The spectral shape of the noise emitted is similar to that for a free jet, being centered around a peak frequency f_p

$$f_p = \frac{0.2 M_j c_0}{D_j} \quad M_j < \sqrt{2},\ c_0 = \text{sonic velocity in vena contracta}$$

$$f_p = \frac{0.28 M_j c_0}{D_j \sqrt{M_j^2 - 1}} \quad M_j \ge \sqrt{2}$$

The noise radiated through the pipe walls is influenced by the frequency-dependent wall transmission loss (see Section 8.1).

5.2. Guidelines for Reduced-Noise Design

Noise of constrained jets and flow restrictions is reduced using general approaches described below. Noise reduction techniques for control valves are discussed in several references[14,15, 16,17].

Ultimately, these techniques relate to reducing the mechanical energy in the gas, the flow through a restriction, upstream turbulence, and the propagation of generated sound waves along the pipe or through the pipe into the environment.

5.2.1. Control Valves

- ***Multi-port resistance plates*** (also called diffusers) are appropriate for large pressure drops where a small control range is required. The plate should be sized for the maximum flow condition with the control valve 100% open. The control valve is then sized to be 30% or more open at minimum flow. Noise reduction of 15 dB(A) is achievable for a fixed control point. The benefit is reduced for greater departures from the maximum flow condition.
- ***Valve Trim:*** Some forms of **valve trim** provide special flow control elements (e.g., a series of perforated disks) whose purpose it is to provide pressure drop in stages. More gradual deceleration reduces the pressure recovery. Check with manufacturers regarding the availability of valve trim for the control valve in question: it may not be available for all valve types and sizes and is often difficult to install in retrofit situations. An example of valve trim is depicted in Figure 4. Although this particular trim is intended for liquid service, it demonstrates the principles clearly.

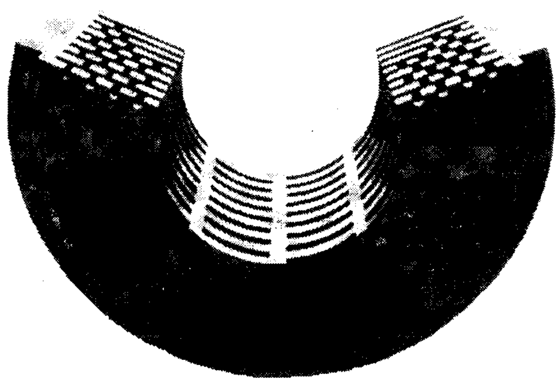

Figure 4: An Example of Valve Trim

(Masoneilan/Dresser)

- ***Apply lagging*** to the exterior of the pipe. Focus on the area downstream of the valve.
- ***Multiple flow paths:*** If control over a wide range is required, consider using multiple control valves mounted in parallel, each sized for a different control range.
- Use ***straight pipe runs*** of least 6 pipe diameters between control valve and fittings both upstream and downstream. Design for minimum pressure drop at fittings. Avoid sudden expansions and contractions in general. At pipe junctions, wyes and tees, use gradual (large radius) transitions wherever possible. Replace tees with wyes whenever possible. In general, the fittings with lowest pressure drop will produce the least noise.
- ***Select the smallest diameter valve*** that will carry and control the maximum flow expected.
- ***Avoid anomalous flow conditions***: avoid operating a valve at less than 30% of its rated capacity.
- ***Use valves with high values of F_L*** near full capacity. A given valve typically has better pressure recovery performance near full capacity than at minimum capacity.
- ***Special low-noise valves*** incorporate high values of F_L and, in some cases, built-in valve trim. A noise reduction benefit of 15 to 25 dB(A) is achievable with proper selection.
- ***Install an in-line silencer:*** The effectiveness of an in-line silencer is estimated at 20 dB(A). This applies to both noise within the pipe and noise radiated from the pipe up- or downstream of the silencer. Typical practice is to place the silencer downstream of the valve. Experience has shown that a downstream silencer alone brings a benefit of only about 10 dB(A) in some cases because the sound upstream of the valve remains unattenuated. In order to realize the full 20 dB(A) benefit of the silencer, both upstream and downstream silencers may be necessary. The piping between the valve and silencer should be selected with thick walls and perhaps be covered with lagging.
- ***Increase wall thickness of pipe***. Doubling the pipe wall thickness could bring a 5 dB(A) reduction.
- ***Coordinate f_p and pipe TL:*** Select valve diameter, pipe diameter and thickness so that the peak frequency f_p is several times greater than f_0, and preferably greater than f_r. Failing this, f_p should be less than f_c. Avoid selecting f_p similar to f_c.

- ***Use pressure reducing plates*** or valve trim to create smaller jets, thereby increasing peak frequency f_p relative to f_r. Smaller jet diameters usually take better advantage of pipe transmission loss by increasing f_p away from f_c[iii].

5.2.2. Measurement Orifices and Venturis

- ***Reduce pressure drop***: the measurement orifice or venturi with lowest pressure drop is desired. This means maximizing the diameter ratio A_O/A_i and using gradual inlet and discharge angles for venturis.
- ***Use straight pipe runs*** of least 6 pipe diameters between control valve and fittings both upstream and downstream. Note that reducing large-scale turbulence in this manner is also important for measurement accuracy.
- ***Apply lagging*** to the exterior of the pipe. Focus on the area downstream of the valve.

5.2.3. Vacuum Vents

- A series of ***pressure-reducing plates*** may be considered.
- ***Use a well-rounded inlet.*** Avoid obstructions or sharp edges in or near the throat.
- In vacuum blowdown applications, ***lengthen the blowdown time*** by reducing the mass flow rate.
- ***Apply lagging*** to the exterior of the pipe.

5.3. Structural Fatigue Criterion

High sound levels and the accompanying vibration make structural fatigue of valve parts a possibility. Valve manufacturers recommend that valve noise at 1 meter from the pipe wall be limited to 115 to 120 dB(A) to avoid fatigue. Note that in-line silencers or lagging are not helpful at reducing vibration levels within the valve where the danger of fatigue is greatest. A more detailed discussion of structural fatigue is presented in Section 8.1.3, page 8-3.

The Control Valve spreadsheet calculates a structural fatigue criterion based on sound power level within the pipe and compares it to computed in-pipe conditions. If interior sound levels are within 10 dB of Structural Fatigue Criterion, design alternatives that reduce noise at the source should be considered.

[iii] In cases where f_c is less than f_p, the addition of valve trim may be detrimental.

5.4. Noise Emission Estimation Using Workbook

Spreadsheets:

- Control Valves
- Orifices, Venturis and Vacuum Vents

Required Inputs:

- General: Gas Compressibility Factor Z, mass flow rate m'
- Upstream conditions: pressure P_1, temperature T_1,
- Valve, Piping and Nozzle: valve coefficient C_V, valve diameter D_v, downstream pipe diameter D_D, upstream pipe diameter D_U, pipe wall thickness t_p, orifice or venturi outer diameter D_o, orifice or venturi inner diameter D_i
- Downstream conditions: pressure P_2, temperature T_2,
- Observer: distance r, angle θ

Notes:

- The Spreadsheet performs a rudimentary valve sizing algorithm for gases. Select valve type using the scrolling box in Line 2a (note that the same type may be listed several times for various service conditions). The C_V and D_V of the valve is estimated. The user must enter the actual C_V selected. Consult valve manufacturers for greater accuracy in sizing.
- Sound power levels internal to the pipe are compared to the structural fatigue limits for the given pipe diameter in Part 4.
- The user may elect the inclusion of various control-valve related noise control elements, including valve trim, in-line silencers upstream and/or downstream of the valve, and downstream resistance plates. Note that to use the in-line silencer selection here it is not necessary to refer to the Silencers spreadsheet (See Section 8.4). The silencer performance used here is generic.

5.5. Noise Emission Equations for Control Valve Noise

The predictive equations for noise emission below follow the approach of Baumann[18] as adapted by Bies and Hansen[19] and Beranek and Ver[17]. A similar approach is adopted in various standards[20,21]. The user should be aware that most valve manufacturers incorporate noise prediction into their sizing software.

The overall sound pressure level inside the pipe is estimated as:

$$L_{pi,overall} = -56 + 10\log\left(\frac{\eta C_v F_L P_1 P_2 c_0^{\ 4} G^2}{D_v^{\ 2}}\right)$$

$$F_L^{\ 2} = \frac{P_1 - P_2}{P_1 - P_0}$$

where C_V is in customary units (gals/min per psia$^{1/2}$), η, F_L and G are dimensionless, P_1, P_2 and P_0 (the pressure in the vena contracta) are in newtons/meter^2 and D_v is in meters. The ratio C_V/D_V^2 (where D_V is in millimeters), pressure recovery coefficient F_L, and valve style modifier F_D are tabulated below in Table 5 (page 5-8). The efficiency of conversion η depends on the stream Mach number M_j, as shown below in Figure 5.

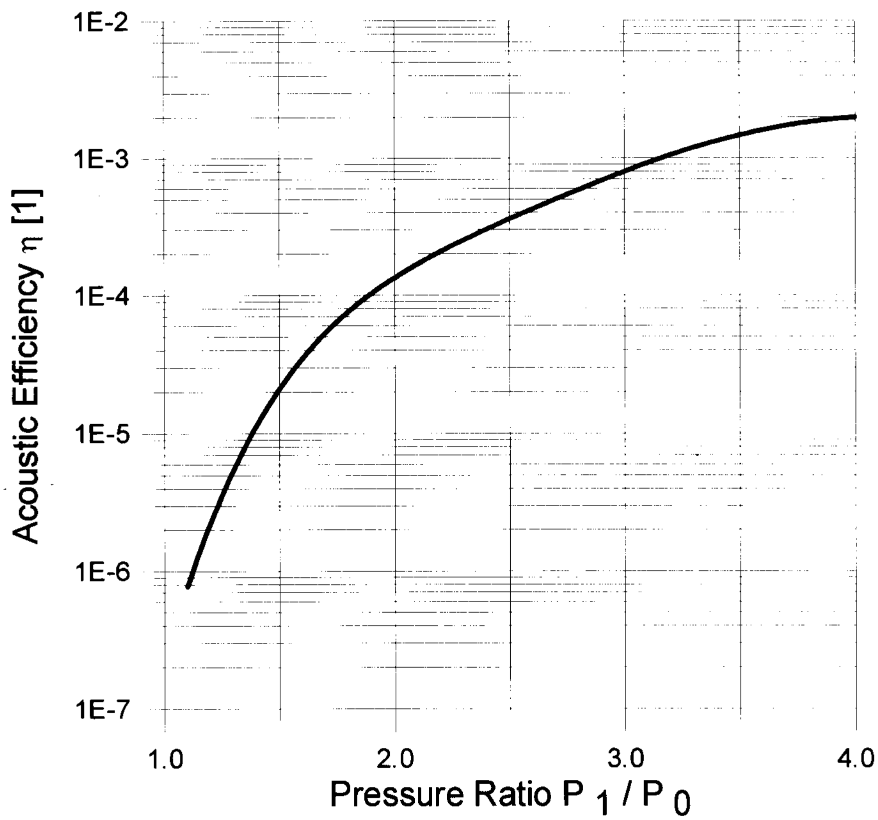

Figure 5: Acoustic Conversion Efficiency vs. Mach Number

Table 5: Typical Constants Associated with Control Valves

Type	Flow To	% Travel	C_V/D_V^2*	F_L	F_D
Globe, single-port parabolic plug	Open	100%	0.020	0.90	0.46
Globe. Single-port parabolic plug	Open	75%	0.015	0.90	0.36
Globe. Single-port parabolic plug	Open	50%	0.010	0.90	0.28
Globe. Single-port parabolic plug	Open	25%	0.005	0.90	0.16
Globe. Single-port parabolic plug	Open	10%	0.002	0.90	0.10
Globe. Single-port parabolic plug	Close	100%	0.025	0.80	1.00
Globe, V-port plug	Open	100%	0.016	0.92	0.50
Globe, V-port plug	Open	50%	0.008	0.95	0.42
Globe, V-port plug	Open	30%	0.005	0.95	0.41
Globe, four-port cage	Open	100%	0.025	0.90	0.43
Globe, four-port cage	Open	50%	0.013	0.90	0.36
Globe, six-port cage	Open	100%	0.025	0.90	0.32
Globe, six-port cage	Open	50%	0.013	0.90	0.25
Butterfly valve, swing-through vane	N/A	75° open	0.050	0.56	0.57
Butterfly valve, swing-through vane	N/A	60° open	0.030	0.67	0.50
Butterfly valve, swing-through vane	N/A	50° open	0.016	0.74	0.42
Butterfly valve, swing-through vane	N/A	40° open	0.010	0.78	0.34
Butterfly valve, swing-through vane	N/A	30° open	0.005	0.80	0.26
Butterfly valve, fluted vane	N/A	75° open	0.040	0.70	0.30
Butterfly valve, fluted vane	N/A	50° open	0.013	0.76	0.19
Butterfly valve, fluted vane	N/A	30° open	0.007	0.82	0.08
Eccentric rotary plug valve	Open	50° open	0.020	0.85	0.42
Eccentric rotary plug valve	Open	30° open	0.013	0.91	0.30
Eccentric rotary plug valve	Close	50° open	0.021	0.68	0.45
Eccentric rotary plug valve	Close	30° open	0.013	0.88	0.30
Ball valve, segmented	Open	60° open	0.018	0.66	0.75
Ball valve, segmented	Open	30° open	0.005	0.82	0.63

The peak frequency f_p of the noise spectrum depends on the velocity of the flow and the diameter of the jet D_j as

$$f_p = \frac{0.2 M_j c_0}{D_j} \qquad M_j < \sqrt{2}$$

$$f_p = \frac{0.28 c_0}{D_j \sqrt{M_j^2 - 1}} \qquad M_j \geq \sqrt{2}$$

where M_j is the Mach Number of the flow in the jet, D_j is the diameter of the jet (not the valve body or the pipe), and c_0 is the sonic velocity in the vena contracta.

The jet diameter may be estimated as

$$D_j \approx 4.6 \times 10^{-3} F_d \sqrt{C_V F_L}$$

where F_d is termed the "valve style modifier", empirically determined, and tabulated above in Table 5.

The stream Mach Number M_j is calculated as follows:

$$M_j = \left[\frac{2}{\gamma - 1} \left(\left(\phi \frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \right]^{\frac{1}{2}}$$

where

$$\phi = \left(\frac{\gamma+1}{2} \right)^{\frac{\gamma}{\gamma-1}} - F_L^2 \left(\left(\frac{\gamma+1}{2} \right)^{\frac{\gamma}{\gamma-1}} - 1 \right)$$

The level L_{pi} of the one-third octave band containing the spectrum peak frequency is $L_p = L_{pi,overall} - 8$. For frequencies greater than the peak frequency the spectrum rolls off at the rate of 3.5 dB per octave. For frequencies less than the peak frequency the spectrum rolls off at the rate of 5 dB per octave for the first two octaves and then at the rate of 3 dB per octave at lower frequencies.

The one-third octave band sound pressure levels external to the pipe at one meter from the pipe centerline are calculated as:

$$L_p \Big|_{1m} = L_{pi} - TL + 5 + L_g$$

where

$$L_g = -16 \log \left(1 - \frac{1.3 \times 10^{-5} P_1 C_V F_L}{D_i^2 P_2} \right)$$

and TL is the pipe wall transmission loss. Circular pipe is generally quite effective at blocking the transmission of sound energy at both high and low frequencies. The weakest Transmission Loss occurs at an intermediate frequency, the first mode cut-on frequency of the pipe:

$$f_c = \frac{0.586 c_2}{D_p}$$

where c_2 is the sonic velocity downstream of the valve and D_p is the internal diameter of the pipe.

Best results for noise emission through the pipe wall are obtained when the peak noise emission frequency f_p and the first mode frequency f_c are widely spaced. A more thorough discussion of pipe wall transmission loss (TL) is given below in Section 8.1.

5.5.1. Noise Emission Equations for Measurement Orifices

From a noise emission standpoint, a measurement orifice can be treated as a special case of a control valve. It is essentially a high recovery valve with fixed control position. The values of C_V and F_L obtained below may be substituted into the control valve noise emission equations above.

The value of C_V is estimated from a general relationship with K:

$$C_V = \frac{D[mm]^2}{21.7\sqrt{K}}$$

$$K = \frac{\Delta P}{\frac{1}{2}\rho u^2}$$

K values for sudden contraction and expansion[22] can be estimated as

$$K = 1.53 - 2.58 \, {A_i}/{A_o} + 1.08\left({A_i}/{A_o}\right)^2.$$

where D is the diameter in millimeters at the flow constriction.

Applying isentropic expansion relations and a polynomial curve fit, it can be shown that F_L takes on the following approximate values:

$$F_L = 0.19 + 1.22\frac{A_i}{A_o} - 0.612\left(\frac{A_i}{A_o}\right)^2$$

5.5.2. Venturis

From a noise emission standpoint, a venturi is also similar to a control valve.
For 20° expansions and contractions[3], K can be estimated as

$$K = .82 - 1.18\,{}^{A_i}\!/\!{}_{A_o} + .352\left({}^{A_i}\!/\!{}_{A_o}\right)^2$$

The C_V value is therefore approximately

$$C_V \approx \frac{D[mm]^2}{21.7\sqrt{.82 - 1.18\,{}^{A_i}\!/\!{}_{A_o} + .352\left({}^{A_i}\!/\!{}_{A_o}\right)^2}}$$

where D is the diameter in millimeters at the flow constriction. From isentropic expansion relations it can be shown that an approximate F_L value is

$$F_L = 1.45\sqrt{1 - \left(\frac{\gamma - 1}{2}\frac{A_i}{A_o} + 1\right)^{-\gamma/\gamma - 1}}$$

when Tap 2 is downstream, or $F_L = 1.000$ when Tap 2 is at the vena contracta.

5.5.3. Intake (Vacuum) Vent

A high-velocity intake vent opening is also treated as a special case of a control valve[iv]. Gas accelerates into the opening and in many cases reaches sonic velocity. The resulting jet and possible shock waves are constrained within the pipe. The vacuum vent is treated as a low recovery valve with fixed control position. Based on K values for various inlet geometries[22], the effective C_V and F_L have been estimated and are tabulated below in Table 6.

[iv] Note that this analysis refers to the opening itself and not to control valves governing the flow downstream.

Table 6: C_V and F_L factors for Intake Vent by Geometry

Inlet Geometry	C_V	F_L
Well-rounded	$0.20 \times D[mm^2]$	1.0
Slightly-rounded	$0.09 \times D[mm^2]$	0.9
Projecting Pipe, L/D < 0.5	$0.06 \times D[mm^2]$	0.8
Projecting Pipe, L/D = 0.5	$0.05 \times D[mm^2]$	0.7
Projecting Pipe, L/D > 0.5	$0.04 \times D[mm^2]$	0.6

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